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EVALUATING COMPLEX INLET DISTORTION WITH A PARALLEL COMPRESSOR MODEL: PART 2 – APPLICATIONS TO COMPLEX PATTERNS*

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ABSTRACT

This paper is the second paper (Part 2) in a companion set and presents results of computer simulations using a parallel compressor model developed with the extended concepts presented in the previous paper (Part 1). The computer model, constructed using the parallel compressor theory with extensions, has been exercised and compared with test results from several gas turbine engines to demonstrate the usefulness of this simulation technique. Distortion patterns used in the tests were created using distortion-generator devices such as distortion screens. A technique to simplify complex distortion patterns through approximate means is presented. This simplification is implemented based upon the flow physics of the compression system, thus allowing the model to better represent the distortion pattern while providing minimal impact on the simulation output and the comparison to the test results. The usefulness of the extended parallel compressor model is demonstrated from its ease of use, simplicity, and ability for quick turn-around of results. Applying a timely analysis capability using the demonstrated parallel compressor model provides an additional physical understanding of the effects of complex distortion on compression system operation.

INTRODUCTION

Modeling compression systems using parallel compressor theory has been used for the analysis of compression system operability since the 1960s. Parallel compressor models have been traditionally designed and used for the analysis of circumferential distortion effects as a means to evaluate the impact of various inlet flow field disturbances on compressor operation.

Parallel compressor models can provide insight into physical phenomena that may not be understood by test data alone. Models can fill information gaps and extend the range of test results to areas not tested. In addition, once a model has been validated, the model can become a numerical experiment and the analysis engineer can conduct "what-if" studies to determine possible solutions to performance or operability problems. With some empirical extensions to the theory, the parallel compressor theory can be constructed into a computer simulation for analysis of compression system performance and operability with and without inlet distortion present [1].

The usefulness of the extended parallel compressor model is derived from its ease of use, simplicity, and ability for quick turn-around of results. When engine testing is being undertaken, it is often more desirable to have an analysis capability that is easy and quick to use than to have one that is extremely accurate, especially when understanding basic physics is of primary concern during the test operation. Extreme accuracy may require large amounts of computer resources and take days or weeks to compute a single performance point. While this may be acceptable for design, the limitations of high-fidelity simulations make them impractical to use due to the time constraints imposed by the pace of testing. Applying a timely analysis capability using a parallel compressor simulation provides an additional physical understanding of the effects of complex distortion during the testing process when comparing the analytical and test results.

This paper and a companion paper present a modeling concept and application to real systems for analysis of compression system performance and operability based upon the parallel compressor theory. The first paper (Part 1, Ref. 1) provides a review of the parallel compressor concept and discusses extensions to the original theory. The second paper (Part 2) applies parallel compressor modeling concepts to complex pressure distortions, inlet swirl and combinations of pressure and temperature distortion.

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PARALLEL COMPRESSOR CONCEPT

In classical parallel compressor theory, the compression system is divided into circumferential segments or "tubes" that extend axially down the compressor (Figure 1**).** Each segment has the same exit boundary condition, but can have a different inlet total pressure and/or temperature. The exit boundary is the only location where the modeling technique transfers information from one segment to another.

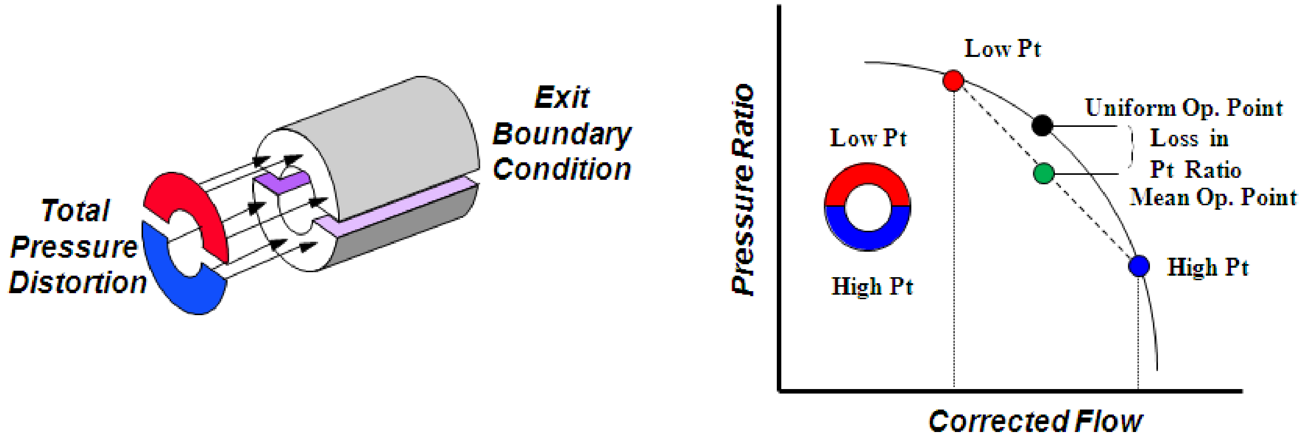

Figure 1. Parallel Compressor Theory Concept [2]

While the inlet conditions to each segment are different, all segments use the same steady-state compressor stage characteristic curves. Each circumferential segment is treated independently with no crossflow between segments. Different levels of pressure or temperature distortion may be imposed upon the inlet, and each segment will operate to its own limit. In this classical form, when one segment reaches the instability limit the entire compression system is considered to be unstable. Using this approach, the mean operating point at instability is a weighted average of the low-flow sector operating at the uniform flow stability boundary, and the high-flow sector operating at some other point far from the stability limit as shown in Figure 1.

Evidence indicates that there is a critical angle of extent for this theory to work (See Ref. 1 for more details). Some investigators suggest that 25-deg is the limit [Ref.1]. ARP-1420 [3] indicates that this is determined by test, but says nothing about how this should be accomplished (mainly because not enough work was done at the time to determine this). It suggests using 25 degrees as a critical angle in the absence of any more information, but doesn't say why. The implication is that it is possible for a small portion of the compressor annulus to operate beyond the observed stability limit provided that there is enough of the annulus operating on the stable side to maintain overall stability. For cases reported within this paper no more than 8-circuferential segments (45-deg extents) were used.

In general, the parallel modeling effort described analyzes the effects of distortion for compression systems with known or calculated clean inlet stage characteristics. The effects of tip clearance or inlet swirl, for instance, would have to be implemented within the stage characteristics or as a modification to the existing known characteristics, and then evaluated with and without inlet distortion.

Model Operational Process

The first step in preparing to use the parallel compression system model is to verify that the model produces appropriate clean inlet overall results. As an example of the this process, the parallel compressor model as applied to the J85 8-stage compression system as reported in Ref. 2 is summarized below.

The parallel compressor model was exercised over a range of corrected speeds and back pressured while at constant corrected speed to a point where system instability was indicated. To obtain a constant corrected speedline with the parallel compressor, the model was set at the desired speed, inlet pressure and temperature while the exit boundary condition was set such that the overall flow rate was near the nominal operating point.

Once steady-state operation was obtained, the exit static pressure was ramped at a rate representing a typical combustor or augmentor fuel pulse, or some other destabilizing event, until overall compressor instability was indicated. Compressor instability is generally determined within the parallel compressor model when a number of stages have reached their instability limit as indicated by the point of zero slope on the stage pressure ratio vs. stage corrected flow input curves.

In the absence of experimental stalling stage data, an engineering judgment was required as to how many stages/segments had to be stalled at time of complete flow breakdown. In general, a majority of stages were required to be stalled to determine compression system instability. This same procedure was followed for inlet distortion with the stalling stage generally found to be in the low pressure region for pressure distortion and in the high temperature region for temperature distortion.

Going Beyond the Design Intent of the Theory

To apply the parallel compressor model to complex distortion goes beyond the design intent of the theory, thus requiring that the analyst understand parallel compressor theory and application of the theory into the elements of the numerical simulation. The user must think in terms of approximating a complex distortion pattern by discretizing over circumferential and radial control volumes that represent the nature of the pattern in that local region. Much of the time this can be accomplished by visual inspection of the pattern. An example of this process is illustrated in Figure 2, which is a representative cruise flight condition distortion pattern for the T-38 trainer aircraft as computed by CFD and reported in Ref. 2

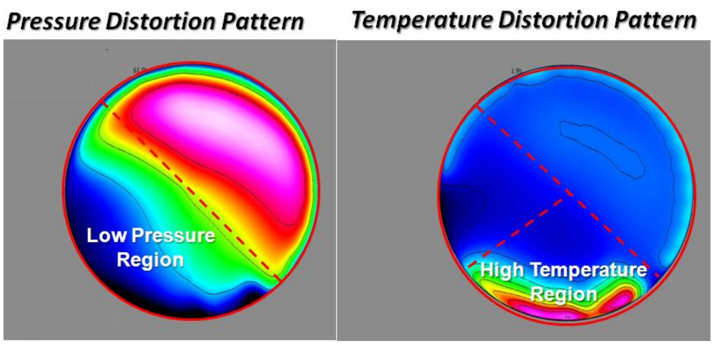

Figure 2. Representative Steady-State CFD AIP Pressure and Temperature Distortion Patterns @ a Typical Cruise Flight Condition [2]

A visual inspection of the CFD pattern at the AIP indicates that the pressure distortion pattern is nearly a 1-per-rev 180-deg circumferential pattern. The temperature pattern is confined to a 90-deg segment near the tip of the machine and in the same quadrant as the pressure distortion. Because of these observations, a simplification of the pattern for use within the parallel compressor model was made as illustrated in Figure 3.

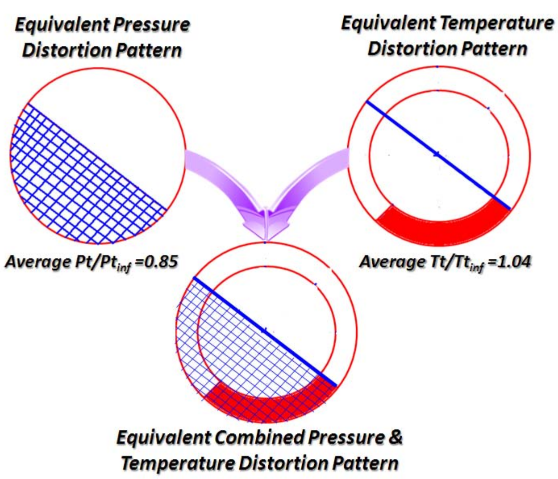

Figure 3. Typical Simplification of Pressure and Temperature Distortion Patterns for Use Within the Parallel Compressor Model [2]

As indicated by the steady-state CFD predictions, a single average pressure defect of 15% was chosen to represent the pressure distortion in the low pressure region. In addition, the temperature distortion was averaged to approximately 4% increase in a 90-deg quadrant out near the tip which was also based upon the CFD results. In addition, these two patterns were made to occur in the same regions (or concurrent), which would be more detrimental than if they were not in the same quadrants.

APPLICATION OF THE PARALLEL COMPRESSOR MODEL TO COMPLEX INLET DISTORTION PATTERNS

This paper will examine five types of distortion issues using several research and military compression system applications:

1. Classical Patterns – Fan A
2. Multiple-Per-Rev – PW308
3. Complex Total Pressure Distortion for Two Military Fans – Fan A and Fan B
4. Combination of Total Pressure and Temperature Distortion – J85-GE-13
5. Flow Angularity or Swirl Distortion – High Tip Speed Compressor, HTSC

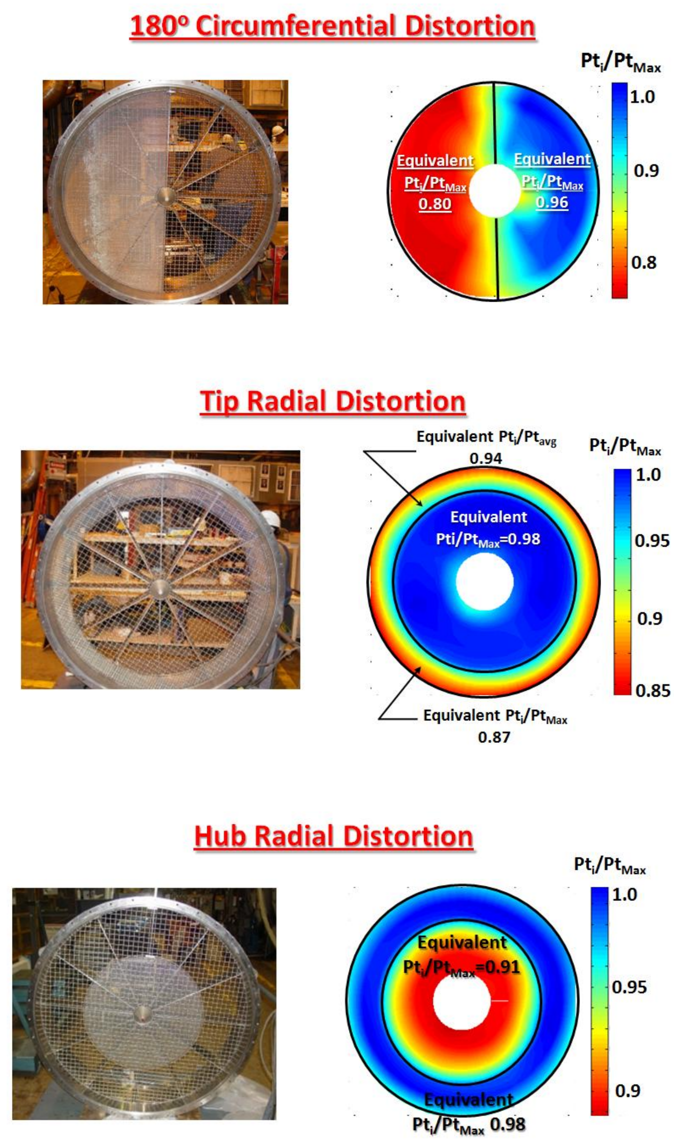

Figure 4. Typical Classical Distortion Screens and Measured Total Pressure Distortion Patterns @ Design Corrected Airflow

In each case, the methodology associated with the approximation of the distortion pattern will be discussed as well as the effect of the complex distortion on the compression system operability characteristics. In all cases, circumferential mass redistribution was implemented as outlined in Part 1 of this two-part paper. Radial mass redistribution was not implemented because it did not prove successful. Implementing a meanline code as a subroutine was not done for any of these cases but has been implemented in a companion paper as applied to swirl distortion [12]

Classical Patterns – Fan A

Classical total pressure distortion in180-deg circumferential, tip radial, and hub radial patterns are used to define compression system sensitivities for use in the SAE ARP-1420 methodology [3]. Typical classical total pressure distortion screens and their associative patterns are presented in Figure 4. Also presented in this figure are representations of the patterns as averages of the sector pressure defect for use within the parallel compressor model. For all model comparisons to experimental results only model prediction to the experimental stability are made. Comparisons to speed lines are not available since the experiment did not hold constant corrected speed during the transient to instability. Speed decreased as the fan was back-pressured to the stability limit.

These classical patterns were applied to a military fan designated as Fan A. A stage-by-stage model was constructed of Fan A and calibrated against clean inlet data and illustrated in Figure 5.

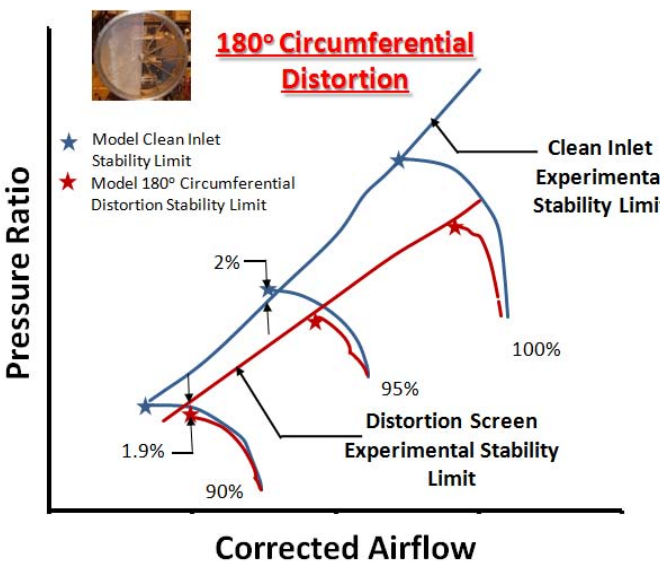

Figure 5. Parallel Compressor Model Prediction of the Loss in Stability Limit for Classical 180-Deg Circumferential Pressure Distortion – Fan A

At three corrected speeds, representations of the classical distortion patterns were implemented as boundary conditions to the parallel compressor model. Comparison to experimental results is presented in Figures 5, 6, and 7 for 180-deg circumferential, tip radial, and hub radial patterns, respectively. As can be discerned visually, comparisons for the circumferential and tip radial predictions were within an

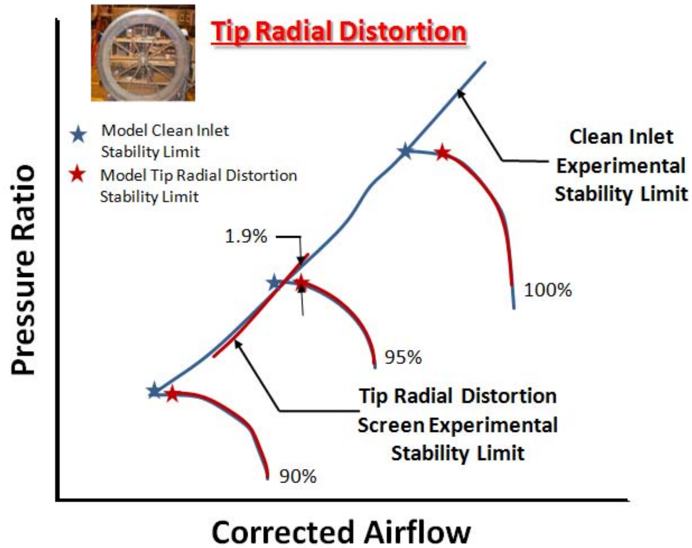

Figure 6. Parallel Compressor Model Prediction of the Loss in Stability Limit for Classical Tip Radial Pressure Distortion – Fan A

acceptable level of accuracy (approx 2% in pressure ratio at a constant corrected airflow). However, in the case of hub radial pattern, a much larger level of error was observed (approximately 6%).

A comment on tip and hub radial distortion effects is in order at this point. Even though the computer modeling technique allows for radial control volumes, the analyst should recognize that since the stage characteristics are generated as an average across all radii, they may not represent what is actually happening at a specified radial location. Thus, if the simulation is applied at the tip location for example; one should not expect that the average stage characteristic would necessarily represent the performance or stalling character that may have been measured in the tip location. There are two caveats to this assumption.

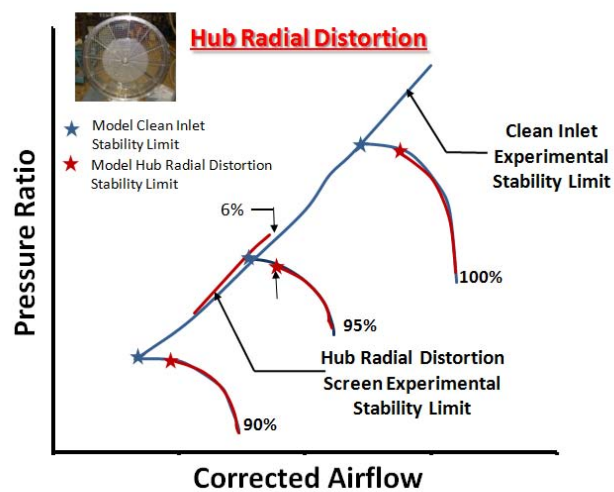

Figure 7. Parallel Compressor Model Prediction of the Loss in Stability Limit for Classical Hub Radial Pressure Distortion – Fan A

The first caveat is that the compressor blade system may not have a large amount of radial variation in pressure or temperature ratio as in the case of a turbojet or high-pressure compressor. In this case, the average will probably represent most radial locations fairly well. The other caveat is that if the compression system typically stalls at the tip location, the average stage characteristic will more closely represent what occurs out at the tip, especially near the stall point. This seems to be the case for Fan A in that the stability limit predicted by the model matches what is observed experimentally within 2%. However, in the hub region, the average characteristic does not accurately represent what happens there since the model prediction misses the stability limit by much as 6%.

Caution is advised because the parallel compressor methodology allows for the discretization of the compression system into radial control volumes which then would use an averaged characteristic. Past attempts to modify those characteristics for radial variation have all failed (Section on Radial Redistribution of Ref. 1). The reader is being advised that for some systems where stall is initiated at the tip, the average characteristic may represent the predicted overall stability limit because the characteristic has tip stall information within its representation. This was true for Fan A (Figure 7).

Multiple-Per-Rev – PW308 [4]

Another type of total pressure distortion is multiple-per-rev, which may have several low-pressure regions that the blades must travel through during a single rotor revolution. A typical scenario for determining system sensitivity is to have 2-4 low pressure regions at nearly equal distance circumferentially to provide the compression system's response to multiple low-pressure regions. One such experimental and parallel modeling example uses the PW308 as reported in Ref. 4.

The distortion test for the PW308 engine was designed to examine several levels of total pressure distortion and several distortion patterns. In the work by Cousins [4], both a 180-deg 1/rev inlet distortion pattern and a 2/rev 90-deg distortion pattern were reviewed. Figure 8 shows the inlet screen and the 2/rev patterns. Only the 2/rev total pressure distortion case as shown in Ref. 4 will be used as an illustration of the parallel compressor model applicability (Figure 9).

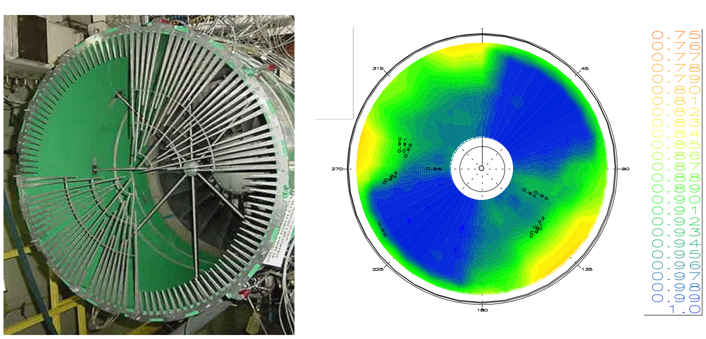

Figure 8. Multiple-Per-Rev Distortion Screen and Measured Total Pressure Distortion Patterns @ Design Corrected Airflow – PW308 [4]

Figure 9 shows the 2/rev measured stability limit on the fan map and the model-predicted speed lines and distorted stability limit.

Figure 10 shows the associated error function. A greater amount of variation is seen in the 2/rev error than in the 1/rev error. It is possible that the radial distortion present in the 2/rev configuration is causing this greater variation. No attempt was made to compensate for the radial distortion present in the 2/rev data. The 103% speed line seemed to match the best, while the other speed line stability limit points were close, some slightly over and some slightly under the measured 2/rev stability limit line. With the screen pattern, it is expected that the circumferential distortion 2/rev pattern would be a greater contributor than the small tip radial at higher speeds, due to the much larger pressure drop of the circumferential distortion. Though, the radial distortion can be defined in the model, the radial mixing model is rather poor, and experience has shown that the representation of radial distortion is not adequate. Therefore, in these simulations it was not used at all.

Every rotor blade has a time constant associated with the time that the rotor takes to respond to the incoming distortion. This

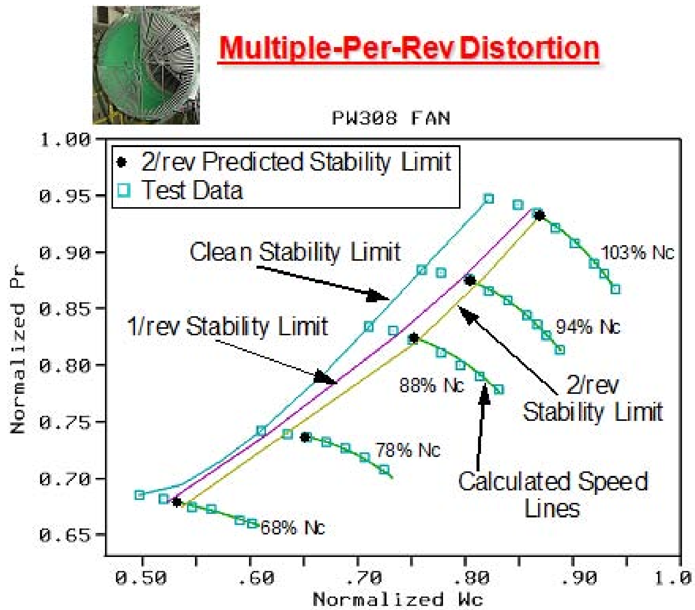

Figure 9. Parallel Compressor Model Prediction of the Loss in Stability Limit for Multiple-Per-Rev Circumferential Pressure Distortion – PW308 [4]

time constant is discussed by Cousins [5] and is represented in the model with a similar concept. The rotor time constant for the PW308 fan is shorter at high fan speeds (where the relative velocity into the fan is higher). Therefore, the fan is also more responsive to the circumferential 90-deg sectors at the higher fan speeds (and less to the tip radial). This would also cause the prediction of the high fan speed stability limit points to be closer to the measured data, which is consistent with the results.

At low fan speeds, the time required for tip recovery outside the blocked 90-deg sectors is longer. The tip radial would typically have the impact of making it more difficult for the tip to recover from the blocked sector. At lower speeds, with a longer time constant, this effect is greater and recovery and tip instability is most likely aggravated.

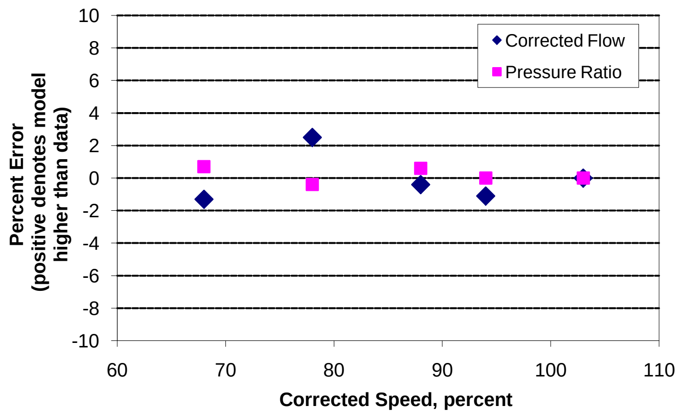

Figure 10. Parallel Compressor Model Prediction of the Loss in Stability Limit for Multiple-Per-Rev Circumferential Pressure Distortion – PW308 [4]

Complex Total Pressure Distortion for Two Military Fans

Complex total pressure distortion refers generally to a realistic pattern generated in a real inlet system measured by dynamic pressure probes as described by the ARP-1420 methodology [3]. These complex patterns are generally simulated by a distortion screen that is made up of various levels of porosity screens and laid out in a pattern that best matches the total pressure profile measure in either a wind tunnel model or actual flight test of the full aircraft-propulsion system. Two examples of complex total pressure distortion screens/patterns are presented in this section, and their effect on compression system stability are evaluated using the parallel compressor model and compared to experimental results. In both cases, some simplification to the pattern was made to represent the complex pattern and then that simplified pattern was used in the parallel compressor model analysis.

Fan A

The complex distortion screen and associated pattern utilized with Fan A is presented in Figure 11. In general, there is more circumferential distortion than there is radial distortion. This allows for a simplification, as shown in Figure 11 that represents the complex pattern with varying levels of circumferential distortion and neglects the radial content. As can be seen in Figure 12, the parallel compressor model predicted the stability limit within 2% of that measured experimentally. This should not be unexpected because the complex pattern has much circumferential distortion associated with it and thus lends itself to parallel compressor analysis. The analysis of the model results (stalling stage segments within the model printed output) indicates that the large 180-deg circumferential sector dominated the instability process again as one might expect when viewing the screen or pattern.

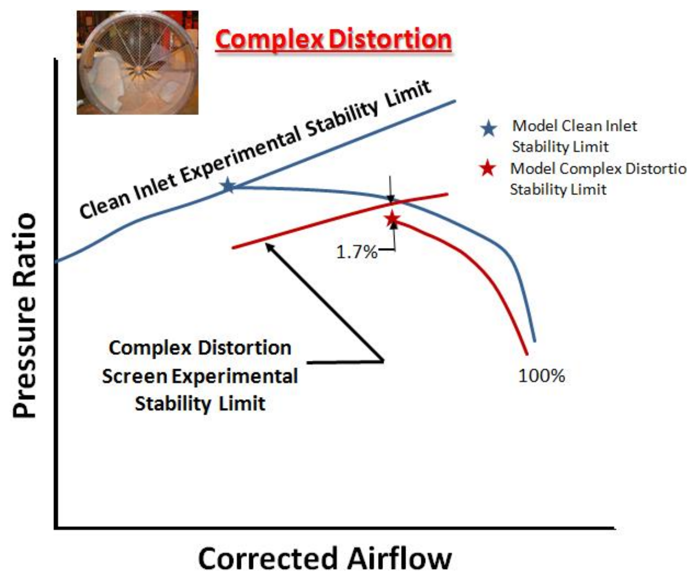

Figure 12. Parallel Compressor Model Prediction of the Loss in Stability Limit for Complex Pressure Distortion – Fan A

Fan B

The complex distortion screen and associated pattern utilized with Fan B is presented in Figure 13. In general, there is more circumferential distortion in the 90-deg sector than there is radial distortion. This allows for a simplification as shown in Figure 13 that represents the complex pattern with varying levels of

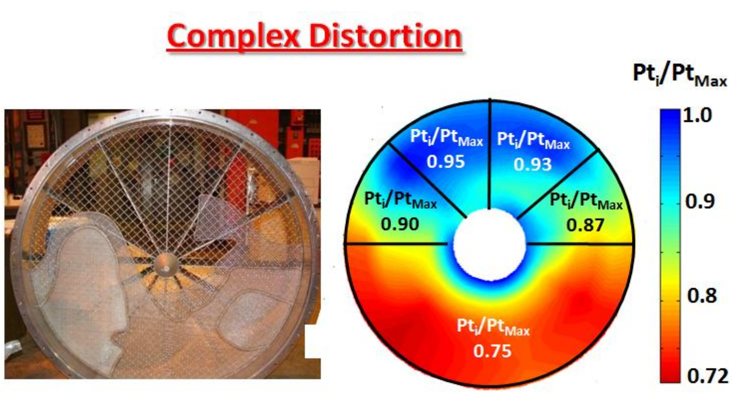

Figure 11. Complex Distortion Screen and Measured Total Pressure Distortion Pattern @ Design Corrected Airflow for Fan A

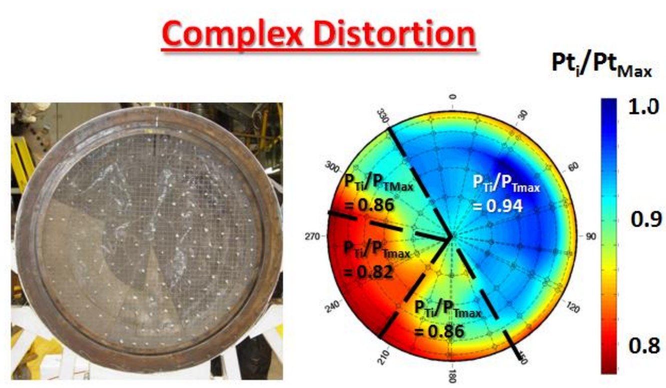

Figure 13. Complex Distortion Screen and Measured Total Pressure Distortion Pattern @ Design Corrected Airflow for Fan B

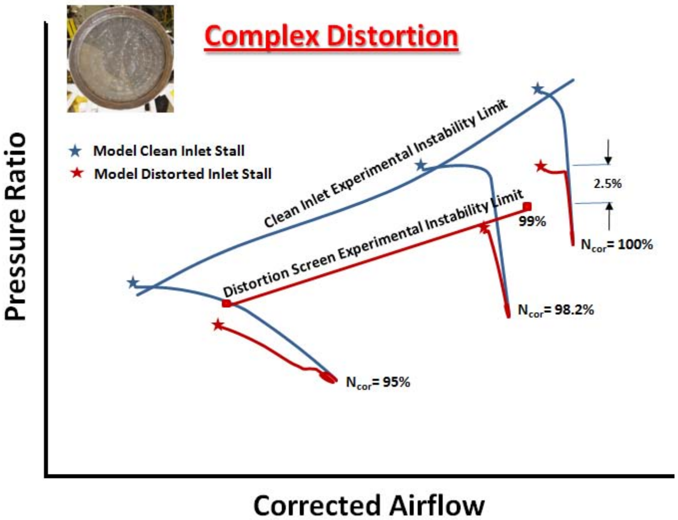

Figure 14. Parallel Compressor Model Prediction of the Loss in Stability Limit for Complex Pressure Distortion – Fan B

circumferential distortion and neglects the radial content.

As can be seen in Figure 14, the parallel compressor model predicted the stability limit within 2.5% of that measured experimentally at the high speed. At both lower speeds, the error was less than 1%. Again this should not be unexpected because the complex pattern has a lot of circumferential distortion associated with it and thus lends itself to parallel compressor analysis. The analysis of the model results (again from printed output) indicates that the large 90-deg mostly circumferential sector dominated the instability process.

Combination of Total Pressure and Temperature Distortion

This section will highlight the effects of both pressure and temperature distortion on 180-deg circumferential segments using the parallel compressor model. Because the superposition of pressure and temperature distortion is not linear, we will look at cases where the pressure and temperature are in opposite segments or reside concurrently in the same 180-deg segment. This numerical investigation was conducted using the J85-GE-13 turbojet engine and was reported by Davis, in Ref. 2. For the parallel compressor model to predict the effects of combined pressure and temperature distortion, it must be validated against experimental results.

Validation For Combined Pressure and Temperature Distortion

Independent experimental pressure and temperature data on the J85-GE-13 engine [6] were taken in the 1970s that can be used for validation. Using the guidance of experimental results from Ref. 6, the parallel compressor model was validated for the combination. Inherently, when combining both pressure and temperature distortion, one has to consider whether the temperature distortion is concurrent with the pressure distortion or opposite to the pressure distortion. This was investigated experimentally and reported in Ref. 6. Presented in Figure 15 is the case when pressure and temperature are in opposite 180-deg sections. As can be seen, the effect of the temperature and pressure distortion have nearly no effect on the stability pressure ratio limit. The opposite situation, however, does not provide such a benign result [6].

Presented in Figure 16 is the case when pressure and temperature are in the same 180-deg section. As evident in the figure, the individual effects seem to be more severe than the

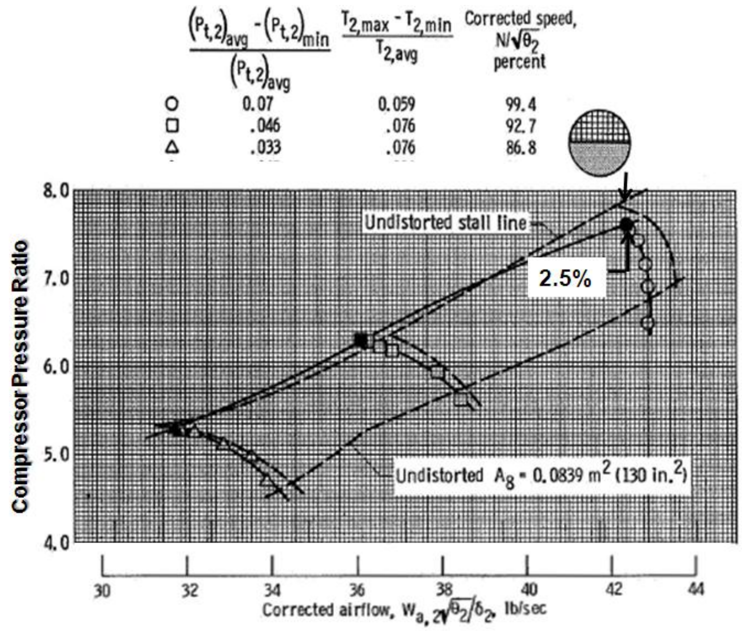

Figure 15. Reproduced Experimental Results for Combined Pressure and Temperature Distortion – Opposed [2]

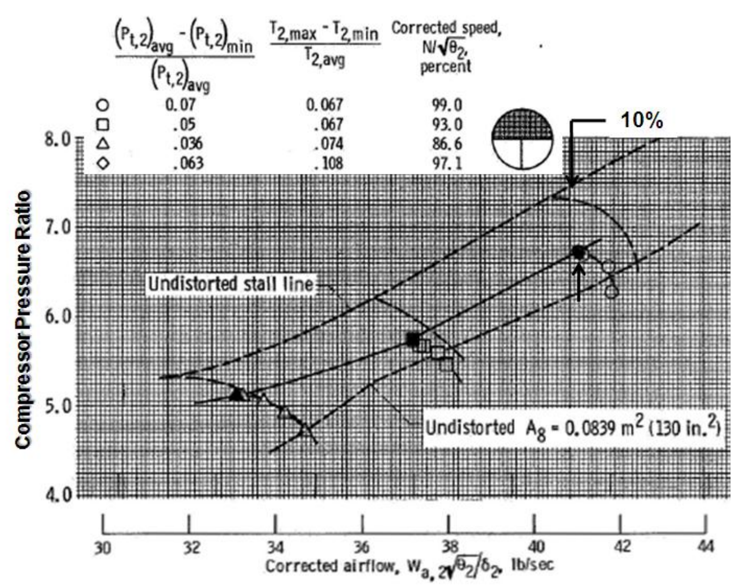

Figure 16. Reproduced Experimental Results for Combined Pressure and Temperature Distortion – Concurrent [2]

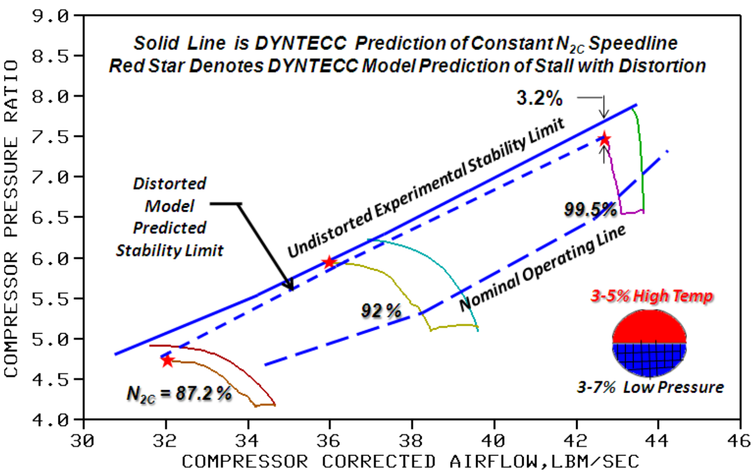

Figure 17. Parallel Compressor Model Results of Temperature and Pressure Distortion – Opposed [2]

linear combination of both effects. These two cases were also executed within the parallel compressor model and were also found to have similar effects (i.e., concurrent is more detrimental than opposed) as presented in Figures 17 and 18.

In summary, the parallel compressor model was validated for the J85-GE-13 eight-stage system. Comparisons of the effect of circumferential pressure and temperature distortions, separately and in combination, agreed well with what had been observed experimentally. Thus, use of the parallel compressor model seems appropriate for more complex distortions especially if they can be tailored to some equivalent form of circumferential distortion.

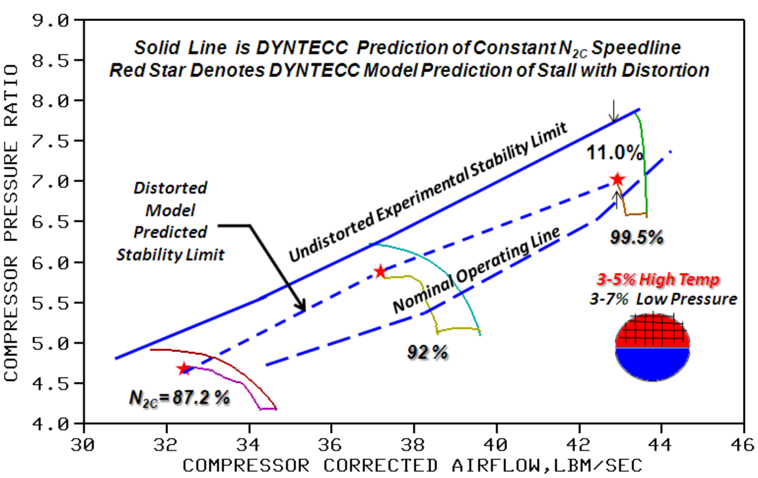

Figure 18. Parallel Compressor Model Results of Temperature and Pressure Distortion – Concurrent [2]

A Specific Complex Pattern of Pressure and Temperature Distortion

Since the initial development of the T-38 Talon trainer, there have been upgrades to both the aircraft and to the J85-GE-5 afterburning turbojet engine to improve takeoff performance, reduce maintenance time and cost, and to decrease fuel consumption. The latest upgrades, referred to as the Propulsion Modernization Program (PMP), focused on improved performance of the T-38's inlets, twin J85-GE-5 afterburning turbojet engines, and improved exhaust nozzle design. The T-38's inlet includes bleed holes upstream of the engine face to provide cooling air flow from the inlet to the engine bay. However, at various locations in the flight envelope, the bay air is pressurized relative to the inlet resulting in reverse flow of hot engine bay air into the inlet. This reverse flow along with inlet heat transfer effects can cause total temperature distortion and reduce engine stability margin. During any flight maneuvers, there will be an associated level of total pressure distortion. When pressure distortion is combined with the temperature distortion due to engine bay flow reversal and inlet heat transfer, losses in stability pressure ratio (or stability margin) may further be increased.

During recent flight tests of the T-38 [2], a series of thermocouples were installed in the inlet of the T-38 just prior to the engine face and after the cooling holes. These thermocouples measured the reverse flow temperature and provided a measure of the temperature distortion present during bay re-ingestion. However, no pressure measurements were obtained and a quantitative assessment of pressure distortion during particular flight maneuvers was not obtained. To approximate what pressure and temperature distortion might be present during certain maneuvers, Reynold's Averaged Navier-Stokes (RANS) steady-state Computational Fluid Dynamics (CFD) calculations were run at two flight conditions using an approximation of the reverse flow from the cooling holes and the new blunter inlet. A representative total pressure and temperature inlet distortion pattern at the Aerodynamic Interface Plane (AIP) is shown in Figure 2 for a typical cruise altitude - Mach number condition.

A visual inspection of the CFD (Figure 2) indicates that the pressure distortion pattern is nearly a 1/rev 180-deg circumferential pattern. The temperature pattern is confined to a 90-deg segment near the tip of the machine and in the same quadrant as the pressure distortion. Because of these observations, a simplification of the pattern for use within the parallel compressor model was made as illustrated in Figure 3.

As indicated by the steady-state CFD predictions, a single average pressure defect of 15% was chosen to represent the pressure distortion in the low-pressure region. In addition, the temperature distortion was averaged to be a single increase in temperature of approximately 4% in a 90-deg quadrant out near the tip which was also based upon the CFD results. If these two patterns were made to occur in the same regions (or concurrently), it would be more detrimental than if they were not in the same quadrants as previously described.

The parallel compressor model was executed with clean inlet, 15% pressure defect for 180-deg inlet total pressure distortion, 4% increase in temperature distortion in the 90-deg tip location, and with both the pressure and temperature distortion present as indicated in Figure 19.

The loss in stability pressure ratio. ΔPRS, is given for comparison for all cases. With 180-deg circumferential pressure distortion of 15% degradation in the low-pressure area, the loss in stability pressure ratio was 6.3%. With 4% increase in temperature in a 90-deg quadrant near the tip, the loss in stability pressure ratio was 3.8%. However, with both temperature and pressure distortion in a concurrent segment, the loss in stability pressure ratio was 10.1% as shown. This particular amount of loss would put the stability limit near the nominal operating line as indicated in Figure 19. Because of temperature distortion, the corrected speed for both the temperature distortion alone and the combination cases are no longer at 100%, but between 97-98% speeds and are shifted to the left of the clean inlet case. This shift in corrected speed is because the mechanical speed was held constant when the

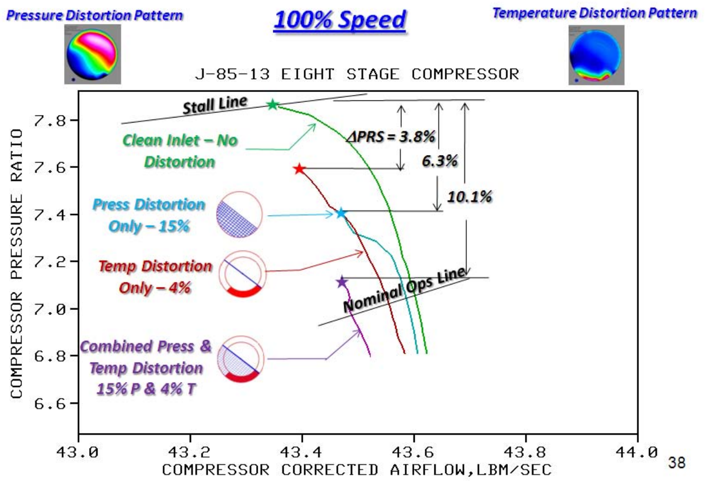

Figure 19. Parallel Compressor Model Prediction of the Loss in Stability Limit for Pressure and Temperature Distortion Separately and in Combination for an Altitude Condition [2]

temperature distortion was present. This provides a clearer picture of what is occurring within the compression system since in general, the control would not have provided a command to change the corrected speed before the system might have gone unstable.

In addition, with pressure distortion alone, the speed line is also shifted to the left as a result of circumferential flow redistribution due to pressure differences between the two parallel segments.

Since there were no experimental results to verify what the model provides, the results stand as a prediction until such time as data are acquired or a similar set of other pattern(s) can be used to verify the model results.

Flow Angularity or Swirl Distortion

Swirl is produced when a flow containing vorticity normal to the flow direction is turned in the plane of the vorticity. Paired swirl is the most common case of swirl and is associated with flow in an S-duct. Low-velocity fluid moves inward in boundary layers at the left and right of the duct (when the turn is in the vertical plane). This results in two vortices rotating in opposite directions at the exit of the turn. When the two vortices have equal magnitude and opposite rotation, this is termed “twin swirl”. Twin swirl has zero circumferential average around the annulus. In the more general case of flow with non-symmetric boundary layers, two vortices are formed of opposite rotation, but different magnitude. A more detailed explanation of swirl can be found in Part I [1] of this two part paper or in Ref. 7.

This application of the parallel compressor model is conducted without any validation of the effect of swirl on system operability. The effect of swirl on each stage of the compression system at present is only a prediction and is presented as an example of how one might approach analysis of swirl on a particular compression system using the parallel compressor modeling approach. Currently, there exists no data for adequate simulation validation.

Development of Stage Characteristics for Swirl Analysis

For the parallel compressor approach to be utilized for the analysis of swirl, stage, blade-row, or overall system maps must be generated that include the effects of swirl. These swirl-maps allow the different sectors of the parallel compressor containing different input swirls to operate at an appropriate operating point on the given off-design speed line. These swirl maps were generated using a meanline code [8] and verified with experimental results for only the non-distorted inlet flow levels reported in Ref. 9.

Thus, based on the sector input swirl and each parallel compressor operating point along a given speed line, the parallel compressor model will find the overall pressure ratio and corrected inlet flow of the composite operating flow and pressure rise. This approach of modeling swirl distortion in a parallel compressor model has been suggested in Refs. 7 and 9. Much of the analysis that is presented within this segment is from Ref. 9 and as such some of the details have been left out as not to duplicate what has already been reported in that reference.

To investigate the effects of swirl on a multi-stage compression system, the High Tip Speed Compressor (HTSC) was chosen since it represented a fan system that has features similar to systems being implemented in today’s military turbofan engines. Again the HTSC has not been run with any form of swirl in front of the compressor and as such the model results are unsubstantiated. However, the HTSC is a two-stage fan, which has been tested extensively at the Compressor Research Facility (CRF) in the 1980s and 1990s under a variety of programs (HTSC, ADLARF, CRFER). In addition, the specific version of the 2-stage fan, HTSC, used in this analysis was configured with no IGV, un-swept rotor blades, and smooth tip casing (Figure 20). Further details of the 2-stage fan can be found in Ref. 10.

The initial application of the meanline code to the HTSC was done to calibrate the meanline code to the clean inlet experimental results at 98.6% speed [10]. Using the meanline code, each flow point was recalculated with an arbitrary inlet flow angle of 5-deg from axial. That is; the flow was given an angular velocity in the direction of rotor rotation (co-swirl) or in the opposite direction of rotor rotation (counter-swirl), thus producing a circumferential swirl component to the inlet flow angularity. Plus or minus five degrees was chosen as a typical angular flow change because that value seemed to provide enough variation in stage performance based upon model results. The loss and deviation correlations used within the meanline code **were not** adjusted as they had been in the clean inlet calibration process but were allowed to reflect the change in both loss and deviation due to the change in inlet flow conditions Swirl mainly affected the first stage performance as shown in Figure 20. The second stage was not affected as shown in Figure 21. The meanline code predictions of stage characteristics for swirl were used as guidelines to develop characteristics for the parallel compressor model.

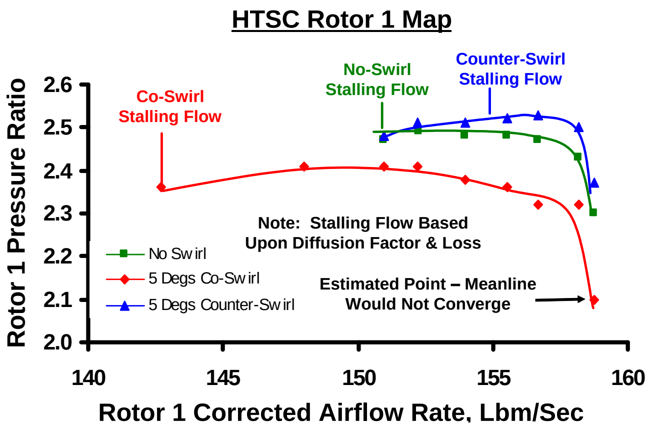

Figure 20. Predictions of +/- 5 Degrees of Swirl on HTSC First Rotor Performance [9]

Since the meanline code can calculate beyond blade row stall, one must evaluate the individual rotor performance and make a determination as to blade row stall and, consequently, system stall. For the clean inlet, the diffusion factor at the experimental stall point was observed to be 0.57, which is near the recommended value of 0.6 [11]. That value and the corresponding loss was used to determine the first blade row stall points for both the co-and counter-swirl cases. Further explanation of this technique can be found in Refs 9 or 12.

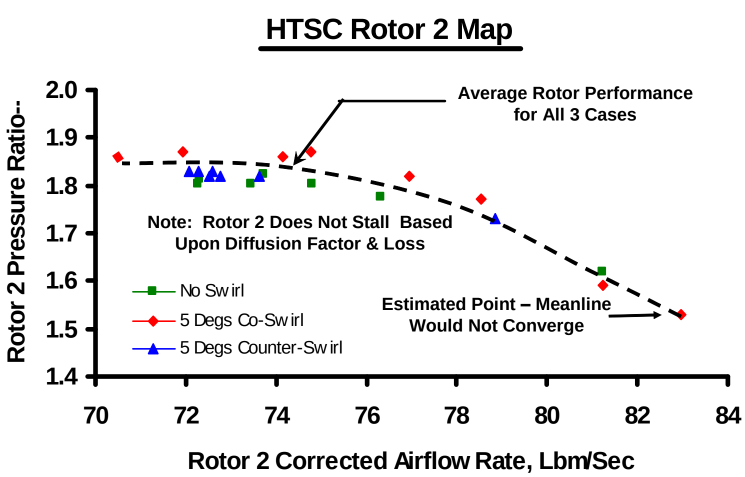

Figure 21. Predictions of +/- 5-Deg of Swirl on HTSC Second Rotor Performance [9]

Implementation of Swirl Stage Characteristics into the Parallel Compressor Model

To simulate the effects of bulk and paired swirl within the parallel compressor model, a simple modification to the affected clean inlet input stage(s) characteristics was performed when swirl was imposed on the inlet control volume. A scale factor on the stage pressure ratio characteristic was applied to simulate what was deduced from the meanline code. Bulk swirl in the co-swirl direction tends to lower the pressure ratio and extend the stalling flow rate, while swirl in the counter-swirl direction tends to increase the pressure ratio and decrease the stalling flow rate. The stall criteria for clean inlet (the flow at which the stage pressure characteristic has a zero slope) was also modified to reflect what was discovered with the meanline code and the diffusion factor investigation when either co-swirl or counter-swirl was present (See Ref. 9).

The results of co-swirl and counter bulk swirl on the performance and operability of the HTSC are presented in Figure 22. Although the results are not identical to those obtained with the meanline code due to the scale factor approach, (Figure 22), they have an appropriate spread in stalling mass flow and a similar range in pressure ratio. Therefore, the scale factor approach described above for creating co-swirl and counter-swirl characteristics within the parallel compressor simulation is adequate for investigating the effects of paired swirl distortions.

The effect of twin-swirl was also investigated with the HTSC and, as can be seen in Figure 23, the twin-swirl performance in terms of pressure ratio is lower than both the no-swirl and counter-swirl bulk swirl cases. In addition, the stall point is between the counter-swirl and co-swirl cases. Examination of the individual sector flow coefficients indicate that the co-swirl stall point controls the overall stall point of the system when subjected to twin swirl and an exit boundary condition of constant static pressure.

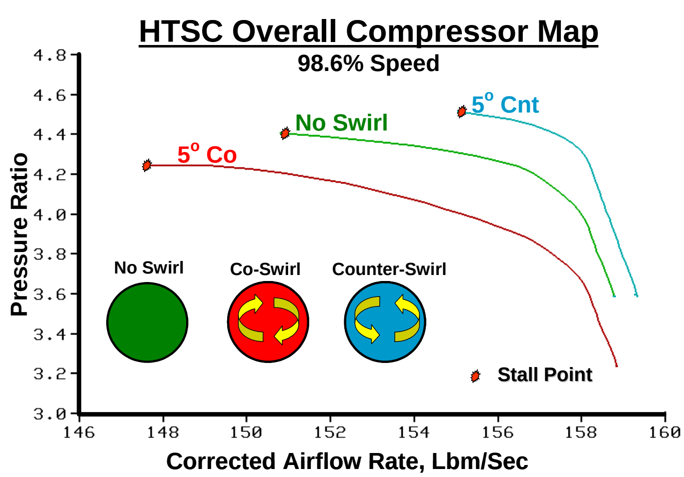

Figure 22. Predictions of Co-and Counter Bulk Swirl on HTSC Performance and Operability [9]

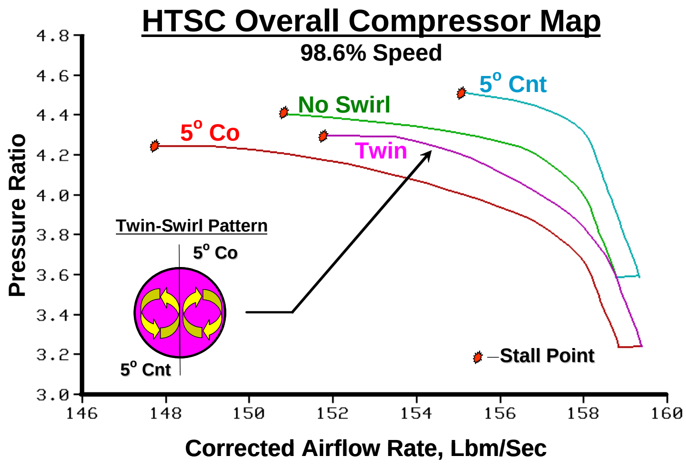

Figure 23. Predictions of Twin-Swirl on HTSC Performance and Operability [9]

CONCLUSIONS

The general intent of the two papers was to provide guidance and insight into how a parallel compressor model might be use and extended to other uses than just circumferential distortion. The reason for this approach was that in the authors experience it has been determined that fast turn-around was more important than super accuracy when involved with engine testing and analysis.

There are many examples of research that is on-going that addresses compression system operability with CFD or Euler modeling techniques. However, these techniques do not provide the quick turn-around necessary in turbine engine test and analysis. Much of these efforts take weeks if not months to set up and produce solutions. Generally when compared to the simpler but much faster empirical techniques the more accurate high fidelity methods only provide a marginally more accurate solution. Many times the same conclusion as to what is happening can be obtained with the simpler approach.

The caveat to this generalization is with radial distortion. Here we have indicated that the parallel compressor model could be used for such investigations but the level of inaccuracies associated with using a bulk stage characteristic (i.e. no radial content) may mask just what the investigator is looking for. In these cases the investigator should use a higher fidelity simulation to quantify the effects of radial redistribution or distortion on system performance and operability. In some cases, when the compression system is known to stall near the tip, the bulk characteristic may reflect an accurate enough picture to drawn some generalized conclusions. However, if hub radial distortion effects are being investigated with a system that stalls near the tip, the parallel compressor solution probably will not produce accurate enough solutions to be effective.

The parallel compressor concept is a versatile concept and can be used for analysis of compressor operability issues beyond those experienced by just circumferential total pressure distortion. This paper has shown that operability analysis with the parallel compressor concept can be value added to the overall analysis process provided that the user is well aware of the limitations associated with using the parallel compressor model and can appropriately interpret the results. The major advantage of using a parallel compressor model is its simplicity, ease of use and rapid turn-around capability. Using parallel compressor results for trending analysis can provide relevant information in a timely manner during an on-going test program.

REFERENCES

1. Cousins, W. T., and Davis, M. W., “Evaluating Complex Inlet Distortion with a Parallel Compressor Model: Part 1 – Concepts, Theory and Limitations”, GT-2011-45067, June 2011.
2. Davis, Milt W., Jr., and Kidman, David, S., “Prediction and Analysis of Inlet Pressure and Temperature Distortion on Engine Operability from a Recent T-38 Flight Test Program”, GT2010-22047, June 2010.
3. SAE S-16 Committee, ARP 1420, Revision B, ”Gas Turbine Inlet Flow Distortion Guidelines,” Society of Automotive Engineers, February 2002.
4. Cousins, W. T., Georges, M. J., and Hooman, R., “Inlet Distortion Testing and Analysis of a High Bypass Ratio Turbofan Engine”, ISABE-2003-1110, 2003.
5. Cousins, W. T., "Characteristics of Stall and Surge Behavior in Axial-Centrifugal Compressors," Ph.D. Dissertation, Virginia Polytechnic Institute and State University, December 1997.
6. Braithwaite, W. M., Graber, E.J., & Mehalic, C., M., “The Effects of Inlet Temperature and Pressure Distortion on TurboJet Performance”, NASA TM X-71431, November 1973.
7. SAE S-16 Committee, AIR 5686, “A Methodology for Assessing Inlet Swirl Distortion”, December 2010.
8. Smith, S. L., “1-D Meanline Code Technique to Calculate Stage-by-Stage Compressor Characteristics,” M.S. Thesis, University of Tennessee, Knoxville, TN, 1999.
9. Davis, Milt, and Hale, Alan, “A Parametric Study on the Effects of Inlet Swirl on Compression System Performance and Operability Using Numerical Simulations”, GT2007-27033, May 2007.
10. Boyer, Keith, “An Improved Streamline Curvature Approach for Off-Design Analysis of Transonic Compression Systems”, Ph.D. Dissertation, Virginia Polytechnic Institute and State University, April 2001.
11. Cumpsty, N. A., **Compressor Aerodynamics**, Longman Scientific & Technical, 1989, pp.47-48.
12. Fredrick, Nicholas and Davis, Milt “Investigation of the Effects of Inlet Swirl on Compressor Performance and Operability Using a Modified Parallel Compressor Model” GT2011-45553, June 2011.